DIESEL ENGINE CHARACTERISTIC, INJECTION, IGNITION, AND FUEL AIR MIXING

NAZARUDDIN SINAGA

Efficiency and Energy Conservation Laboratory
Diponegoro University

Diesel Engine Combustion

1. Characteristics of diesel combustion
2. Different diesel combustion systems
3. Phenomenological model of diesel combustion process
4. Movie of combustion in diesel systems
5. Combustion pictures and planar laser sheet imaging

DIESEL COMBUSTION PROCESS

PROCESS

- Liquid fuel injected into compressed charge
- Fuel evaporates and mixes with the hot air
- Auto-ignition with the rapid burning of the fuel-air that is "premixed" during the ignition delay period
  - Premixed burning is fuel rich
- As more fuel is injected, the combustion is controlled by the rate of diffusion of air into the flame

DIESEL COMBUSTION PROCESS

NATURE OF DIESEL COMBUSTION

- Heterogeneous
  - liquid, vapor and air
  - spatially non-uniform
- turbulent
- diffusion flame
  - High temperature and pressure
  - Mixing limited

The Diesel Engine

- Intake air not throttled
  - Load controlled by the amount of fuel injected
    - >A/F ratio: idle ~ 80
    - >Full load ~19 (less than overall stoichiometric)
- No "end-gas"; avoid the knock problem
  - High compression ratio: better efficiency
- Combustion:
  - Turbulent diffusion flame
  - Overall lean

Diesel as the Most Efficient Power Plant

- **Theoretically, for the same CR, SI engine has higher η_f; but diesel is not limited by knock, therefore it can operate at higher CR and achieves higher η_f**
- **Not throttled - small pumping loss**
- **Overall lean - higher value of γ - higher thermodynamic efficiency**
- **Can operate at low rpm - applicable to very large engines**
  - slow speed, plenty of time for combustion
  - small surface to volume ratio: lower percentage of parasitic losses (heat transfer and friction)
- **Opted for turbo-charging: higher energy density**
  - Reduced parasitic losses (friction and heat transfer) relative to output

Large Diesels: η_f~ 55%
~ 98% ideal efficiency !

Diesel Engine Characteristics (compared to SI engines)

- **Better fuel economy**
  - Overall lean, thermodynamically efficient
  - Large displacement, low speed – lower FMEP
  - Higher CR
    - > CR limited by peak pressure, NOx emissions, combustion and heat transfer loss
  - Turbo-charging not limited by knock: higher BMEP over domain of operation, lower relative losses (friction and heat transfer)
- **Lower Power density**
  - Overall lean: would lead to smaller BMEP
  - Turbocharged: would lead to higher BMEP
    - > not knock limited, but NOx limited
    - > BMEP higher than naturally aspirated SI engine
  - Lower speed: overall power density (P/V_D) not as high as SI engines
- **Emissions: more problematic than SI engine**
  - NOx: needs development of efficient catalyst
  - PM: regenerative and continuous traps

Typical SI and Diesel operating value comparisons

	SI	Diesel
BMEP		
– Naturally aspirated:	10-15 bar	10 bar
– Turbo:	15-25 bar	15-25 bar
Power density		
– Naturally aspirated:	50-70 KW/L	20 KW/L
– Turbo:	70-120 KW/L	40-70 KW/L
Fuel		
– H to C ratio	$CH_{1.87}$	$CH_{1.80}$
– Stoichiometric A/F	14.6	14.5
– Density	0.75 g/cc	0.81 g/cc
– LHV (mass basis)	44 MJ/kg	43 MJ/kg
– LHV (volume basis)	3.30 MJ/L	3.48 MJ/L **(5.5% higher)**
– LHV (CO_2 basis)	13.9 MJ/kgCO_2	13.6 MJ/kgCO_2 **(2.2% lower)**

Disadvantages of Diesel Engines

- Cold start difficulty
- Noisy - sharp pressure rise: cracking noise
- Inherently slower combustion
- Lower power to weight ratio
- Expensive components
- NO_x and particulate matters emissions

Market penetration

- **Diesel driving fuel economy ~ 30% better than SI**
  - 5% from fuel energy/volume
  - 15% from eliminating throttle loss
  - 10% from thermodynamics
    - 2nd law losses (friction and heat transfer)
    - Higher compression ratio
    - Higher specific heat ratio

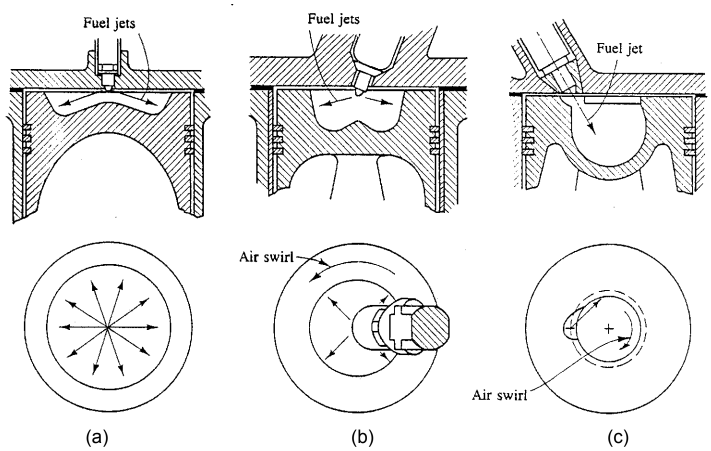
- **Dominant world wide heavy duty applications**
- **Dominant military applications**
- **Significant market share in Europe**
  - Tax structure for fuel and vehicle
- **Small passenger car market fraction in US and Japan**
  - Fuel cost
  - Customer preference
  - Emissions requirement

Applications

- Small (7.5 to 10 cm bore; previously mainly IDI; new ones are high speed DI)
  - passenger cars
- Medium (10 to 20 cm bore; DI)
  - trucks, trains
- Large (30 to 50 cm bore; DI)
  - trains, ships
- Very Large (100 cm bore)
  - stationary power plants, ships

Common Direct-Injection Compression-Ignition Engines

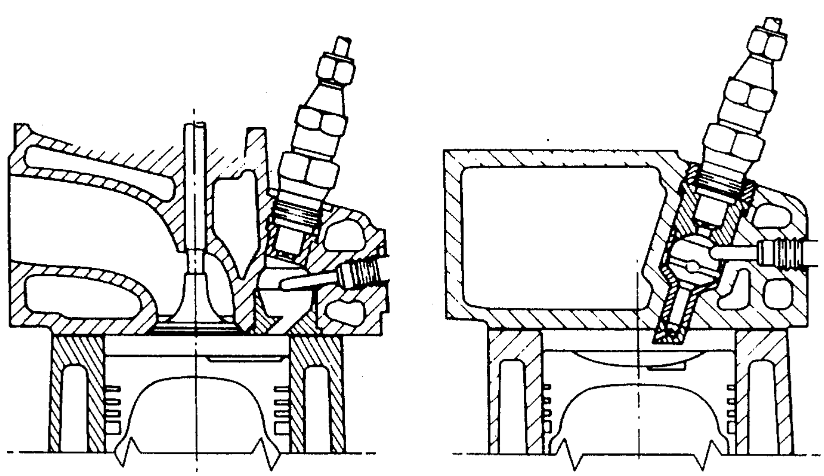
(Fig. 10.1 of text)



(a) Quiescent chamber with multihole nozzle typical of larger engines
(b) Bowl-in-piston chamber with swirl and multihole nozzle; medium to small size engines
(c) Bowl-in-piston chamber with swirl and single-hole nozzle; medium to small size engines

Common types of small Indirect-injection diesel engines

(Fig. 10.2 of text)

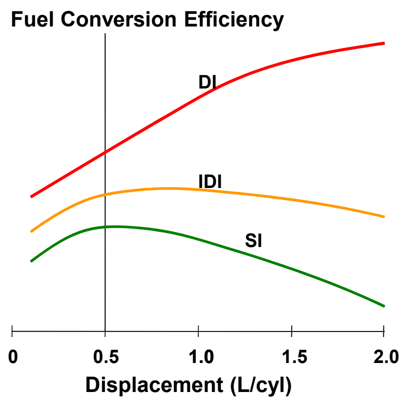


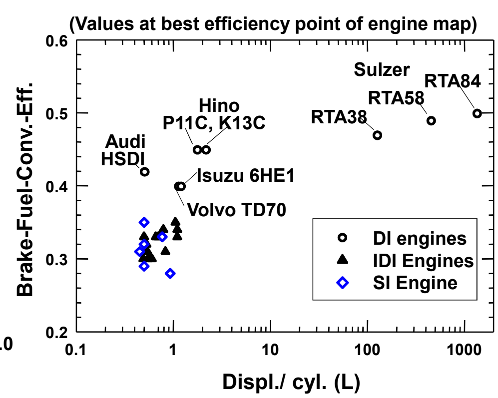
(a) Swirl prechamber **(b) Turbulent prechamber**

Common Diesel Combustion Systems (Table 10.1)

	Direct injection				Indirect injection	
System	Quiescent	Medium swirl	High swirl "M"	High swirl multispray	Swirl chamber	Pre-chamber
Size	Largest	Medium	Medium—smaller	Medium—small	Smallest	Smallest
Cycle	2-/4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke
Turbocharged/supercharged/naturally aspirated	TC/S	TC/NA	TC/NA	NA/TC	NA/TC	NA/TC
Maximum speed, rev/min	120–2100	1800–3500	2500–5000	3500–4300	3600–4800	4500
Bore, mm	900–150	150–100	130–80	100–80	95–70	95–70
Stroke/bore	3.5–1.2	1.3–1.0	1.2–0.9	1.1–0.9	1.1–0.9	1.1–0.9
Compression ratio	12–15	15–16	16–18	16–22	20–24	22–24
Chamber	Open or shallow dish	Bowl-in-piston	Deep bowl-in-piston	Deep bowl-in-piston	Swirl pre-chamber	Single/multi-orifice pre-chamber
Air-flow pattern	Quiescent	Medium swirl	High swirl	Highest swirl	Very high swirl in pre-chamber	Very turbulent in pre-chamber
Number of nozzle holes	Multi	Multi	Single	Multi	Single	Single
Injection pressure	Very high	High	Medium	High	Lowest	Lowest

Effect of Engine Size



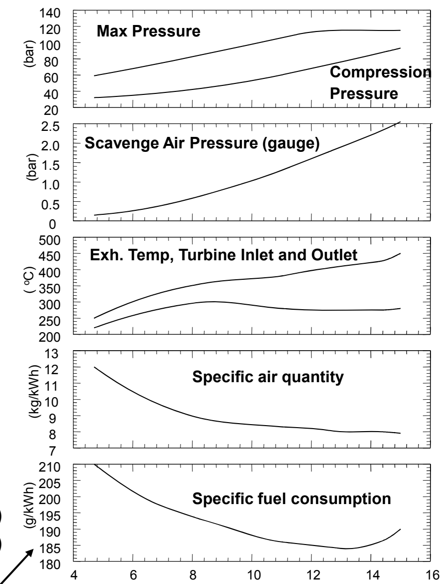


Typical Large Diesel Engine Performance Diagram

Sulzer RLB 90 - MCR 1
Turbo-charged 2-stroke Diesel
- 1.9 m stroke; 0.9 m bore

Rating:
- **Speed: 102 Rev/ min**
  - Piston speed 6.46 m/s
- **BMEP: 14.3 bar**

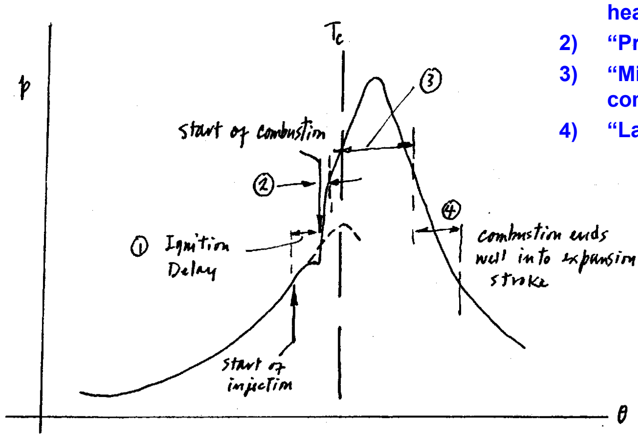
Configurations
- 4 cyl: 11.8 MW (16000 bhp)
- 5 cyl: 14.7 MW (20000 bhp)
- 6 cyl: 17.7 MW (24000 bhp)
- 7 cyl: 20.6 MW (28000 bhp)
- 8 cyl: 23.5 MW (32000 bhp)
- 9 cyl: 26.5 MW (36000 bhp)
- 10 cyl: 29.4 MW (40000 bhp)
- 12 cyl: 35.3 MW (48000 bhp)



$\eta_{f,b}$=0.45 @ 185 g/kW-hr

Sulzer RTA96 engine

Diesel combustion process — direct injection



1) Ignition delay — no significant heat release
2) "Premixed" rapid combustion
3) "Mixing controlled" phase of combustion
4) "Late" combustion phase

Note:
(2) is too fast;
(4) is too slow

Rate of Heat Release in Diesel Combustion
(Fig. 10.8 of Text)

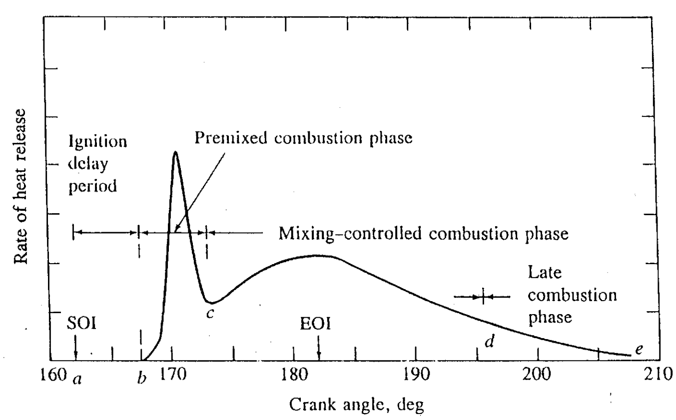
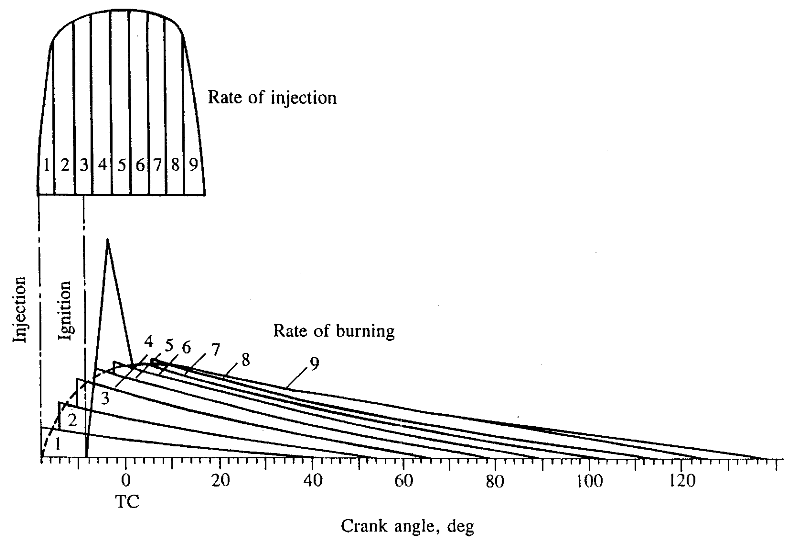
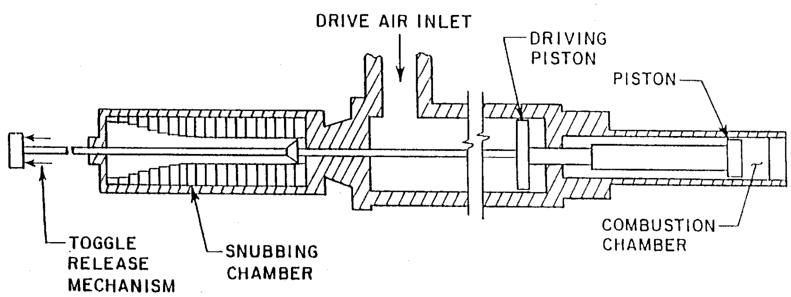


Fig. 10-9

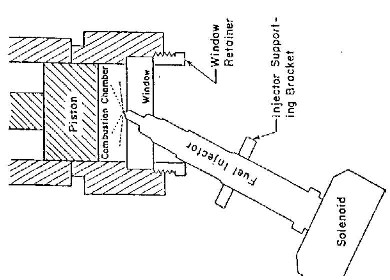
A Simple Diesel Combustion Concept (Fig. 10-8)



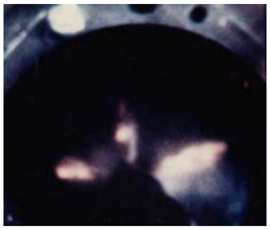
Visualization of Diesel Combustion



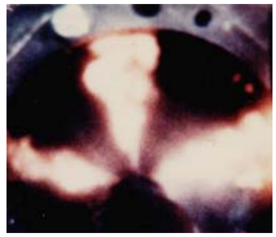
Bore	**10.2 cm**
Stroke	**44.7 cm**
Compression ratio	**15.4**



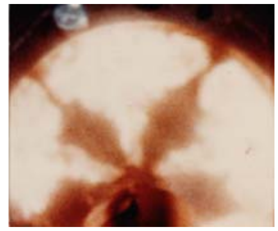
Images From Diesel Combustion



First occurrence of luminous flame
(1.0 ms after start of injection)

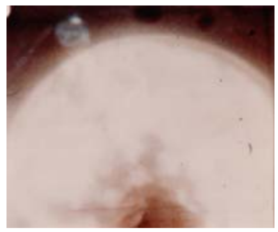


(0.13 ms after ignition)

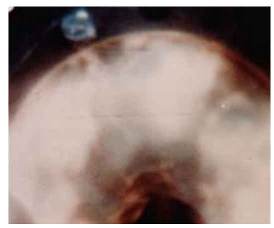


(0.93 ms after ignition)

(1.87 ms after ignition)



End of injection
(2.67 ms after ignition)

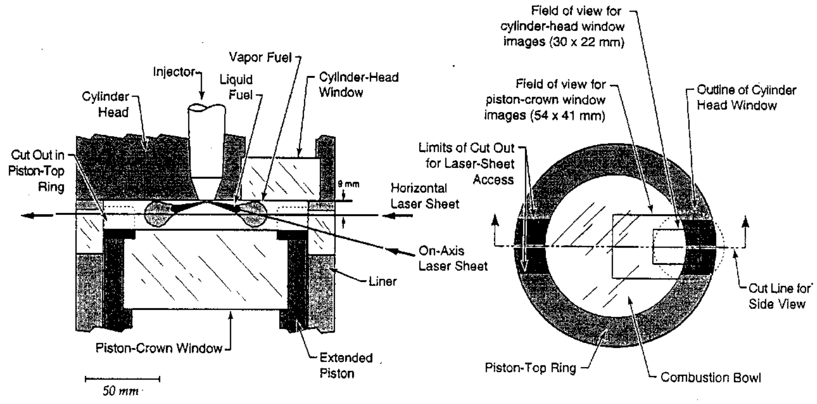


5.33 ms after ignition

FEATURES OF DIESEL COMBUSTION

- **Ignition delay**
  - Auto-ignition in different parts of combustion chamber
- **After ignition, fuel sprays into hot burned gas**
  - Then, evaporation process is fast
- **Major part of combustion controlled by fuel air mixing process**
  - Mixing dominated by flow field formed by fuel jet interacting with combustion chamber walls during injection
- **Highly luminous flame:**
  - Substantial soot formation in the fuel rich zone by pyrolysis, followed by substantial subsequent oxidation

Imaging of Diesel Combustion by Laser Sheet Illumination



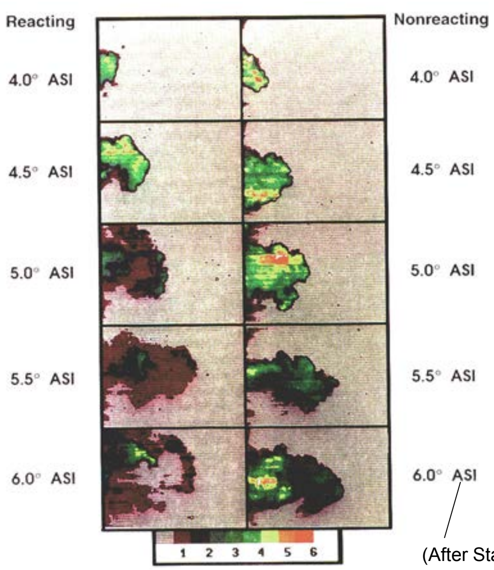
Side View of Combustion Chamber — **Top View of Piston**

Rayleigh scattering
reflection from molecules

Laser Induced Florescence
(pump at) OH @284 nm
PAH @387 nm
NO @ 226 nm

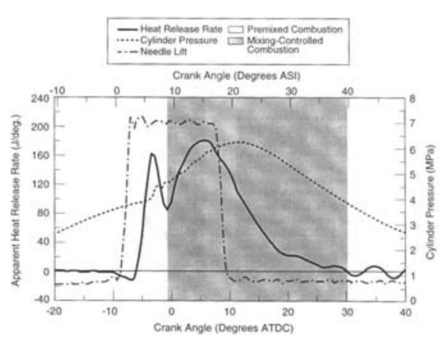
From J.Dec, SAE 970873

Fuel Equivalence Ratio



(After Start of Injection)

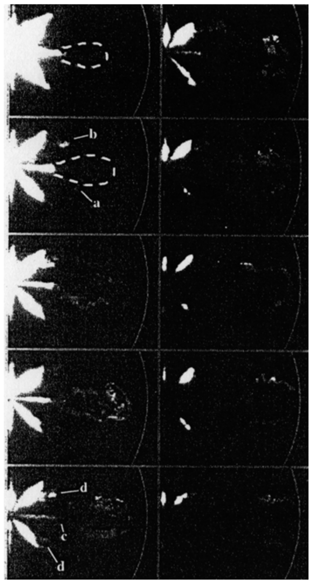
- Obtained by Planar Rayleigh scattering
- Substantial reduction of fuel equivalence ratio in the 'premixed' region indicates fuel-rich oxidation



From J.Dec, SAE 970873

OH Image by PLIF

5.0° ASI
5.5° ASI
6.0° ASI
6.5° ASI
7.0° ASI

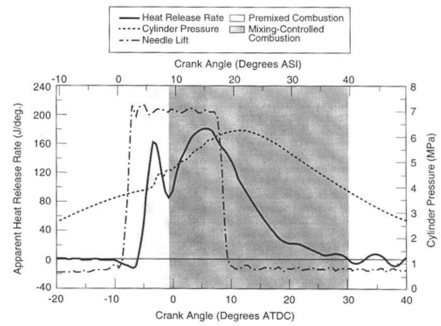


7.5° ASI
8.0° ASI
8.5° ASI
9.0° ASI
9.5° ASI

(Dash lines in the first two frames marks the vapor boundary of the fuel jet)

(After Start of Injection)

Oxidation occurs at the edge of the air and fuel rich region

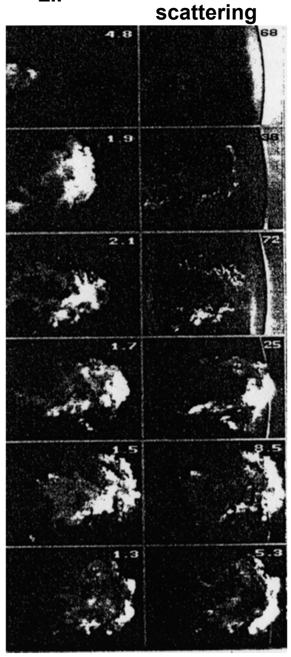


From J.Dec, SAE 970873

Image of the Particulates

LII | Elastic scattering

6.0° ASI
6.5° ASI
7.0° ASI
7.5° ASI
8.0° ASI
8.5° ASI



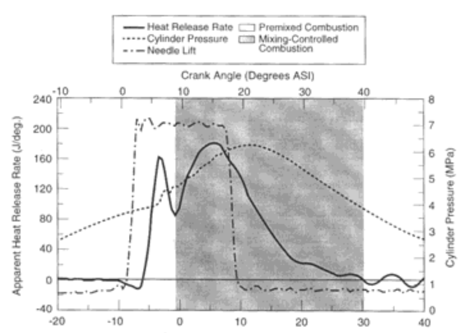
6.0° ASI
6.5° ASI
7.0° ASI
7.5° ASI
8.0° ASI
8.5° ASI

Laser Induced Incandescence

(signal ~ d^3; observe small particles)

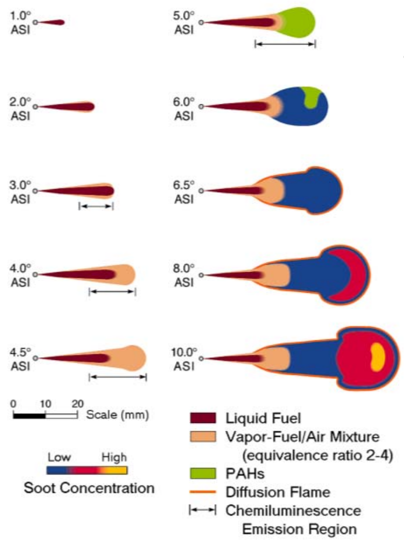
Elastic scattering

(signal ~ d^6; observe large particles)

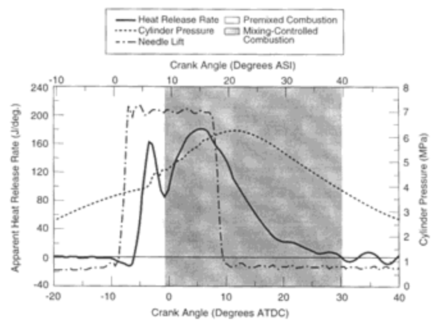


From J.Dec, SAE 970873

Diesel Ignition, Premixed Burning and Transition into Diffusion Burning



- Premixed burning
  - Release of energy from fuel rich combustion
- Diffusion burning
  - Oxidation of incomplete products of the rich premixed combustion and fuel vapor at the 'jet'/ air interface



Figures from J.Dec, SAE 970873

Diesel injection, ignition, and fuel air mixing

1. Fuel spray phenomena
2. Spontaneous ignition
3. Effects of fuel jet and charge motion on mixing-controlled combustion
4. Fuel injection hardware
5. Challenges for diesel combustion

DIESEL FUEL INJECTION

The fuel spray serves multiple purposes:

- Atomization
- Fuel distribution
- Fuel/air mixing

Typical Diesel fuel injector

- Injection pressure: 1000 to 2200 bar
- 5 to 20 holes at ~ 0.12 - 0.2 mm diameter
- Drop size 0.1 to 10 μm
- For best torque, injection starts at about 20º BTDC

Injection strategies for NOx control

- Late injection (inj. starts at around TDC)
- Other control strategies:
  - Pilot and multiple injections, rate shaping, water emulsion

Diesel Fuel Injection System

(A Major cost of the diesel engine)

- Performs fuel metering
- Provides high injection pressure
- Distributes fuel effectively
  - Spray patterns, atomization etc.
- Provides fluid kinetic energy for charge mixing

Typical systems:

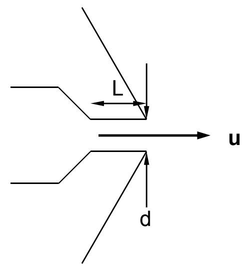
- Pump and distribution system (100 to 1500 bar)
- Common rail system (1000 to 1800 bar)
- Hydraulic pressure amplification
- Unit injectors (1000 to 2200 bar)
- Piezoelectric injectors (1800 bar)
- Electronically controlled

EXAMPLE OF DIESEL INJECTION

(Hino K13C, 6 cylinder, 12.9 L turbo-charged diesel engine, rated at 294KW@2000 rpm)

- Injection pressure = 1400 bar; duration = 40^{o}CA
- BSFC 200 g/KW-hr
- Fuel delivered per cylinder per injection at rated condition
  - 0.163 gm ~0.21 cc (210 mm^3)
- Averaged fuel flow rate during injection
  - 64 mm^3/ms
- 8 nozzle holes, at 0.2 mm diameter
  - Average exit velocity at nozzle ~253 m/s

Typical physical quantities in nozzle flow



- Diesel fuel @ 100°C
  - s.g. ~ 0.78, μ~5x10^{-4} N-s/m^2
- Nozzle diameter ~0.2 mm
- L/d ~ 5 to 10
- Reynolds No. ~ 10^5 (turbulent)
- Pressure drop in nozzle
  ~30 bar << driving pressure (~1000 bar)
- Injection velocity

$$u \approx \sqrt{\frac{2\Delta P}{\rho_{fuel}}} \approx 500 \text{ m/s @ } \Delta P \text{ of 1000 bar}$$

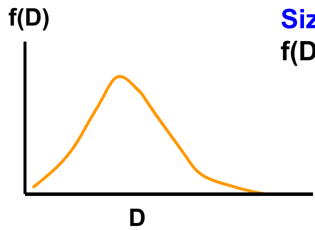
Fuel Atomization Process

- Liquid break up governed by balance between aerodynamic force and surface tension

$$\text{Webber Number } (W_b) = \frac{\rho_{gas} u^2 d}{\sigma}$$

- Critical Webber number: $W_{b,critical}$ ~ 30; diesel fuel surface tension ~ 2.5x10^{-2} N/m
- Typical W_b at nozzle outlet > $W_{b,critical}$; fuel shattered into droplets within ~ one nozzle diameter
- Droplet size distribution in spray depends on further droplet breakup, coalescence and evaporation

Droplet size distribution



Size distribution:

f(D)dD = probability of finding particle with diameter in the range of (D, D + dD)

$$1 = \int_0^\infty f(D)dD$$

Average diameter

$$\overline{D} = \int_0^\infty f(D)\,D\,dD$$

Volume distribution

$$\frac{1}{V}\frac{dV}{dD} = \frac{f(D)\,D^3}{\int_0^\infty f(D)\,D^3 dD}$$

Sauter Mean Diameter (SMD)

$$D_{32} = \frac{\int_0^\infty f(D)\,D^3 dD}{\int_0^\infty f(D)\,D^2 dD}$$

Droplet Size Distribution

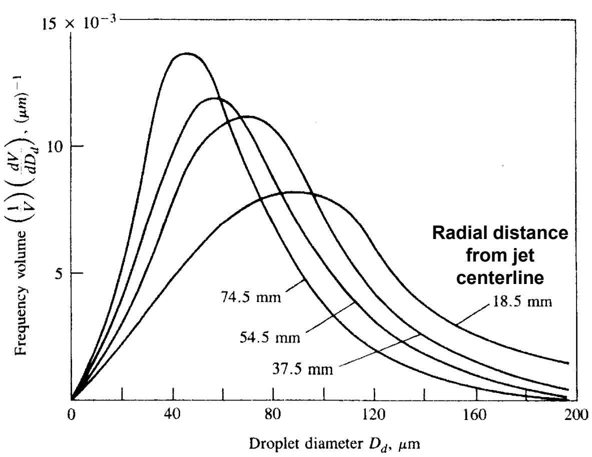
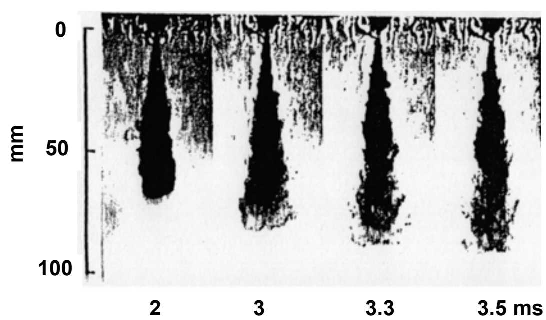


Fig. 10.28 Droplet size distribution measured well downstream; numbers on the curves are radial distances from jet axis. Nozzle opening pressure at 10 MPa; injection into air at 11 bar.

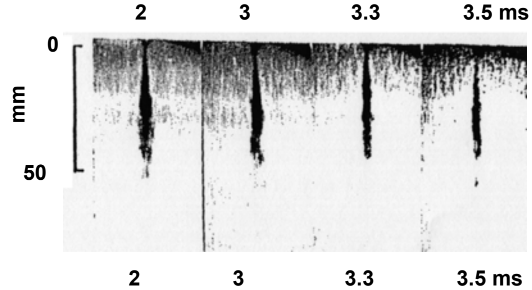
Droplet Behavior in Spray

- Small drops (~ micron size) follow gas stream; large ones do not
  - Relaxation time $\tau \propto d^2$
- Evaporation time $\propto d^2$
  - Evaporation time small once charge is ignited
- Spray angle depends on nozzle geometry and gas density : $\tan(\theta/2) \propto \sqrt{(\rho_{gas}/\rho_{liquid})}$
- Spray penetration depends on injection momentum, mixing with charge air, and droplet evaporation

Spray Penetration: vapor and liquid (Fig. 10-20)



Shadowgraph image showing both liquid and vapor penetration



Back-lit image showing liquid-containing core

Auto-ignition Process

PHYSICAL PROCESSES (Physical Delay)

- Drop atomization
- Evaporation
- Fuel vapor/air mixing

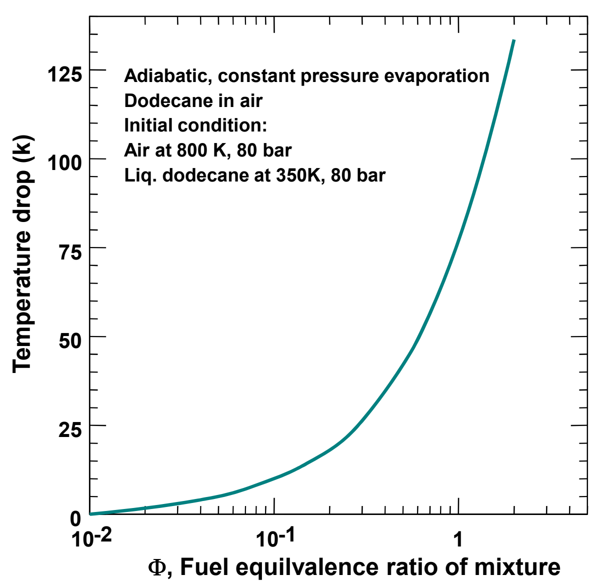
CHEMICAL PROCESSES (Chemical Delay)

- Chain initiation
- Chain propagation
- Branching reactions

CETANE IMPROVERS

- Alkyl Nitrates
  - 0.5% by volume increases CN by ~10

Mixture cooling from heat of vaporization



Ignition Mechanism: similar to SI engine knock

CHAIN BRANCHING EXPLOSION

Chemical reactions lead to increasing number of radicals, which leads to rapidly increasing reaction rates

Chain Initiation

$$RH + O_2 \Rightarrow \dot{R} + H\dot{O}_2$$

Chain Propagation

$$\dot{R} + O_2 \Rightarrow R\dot{O}_2, \text{etc.}$$

Formation of Branching Agents

$$R\dot{O}_2 + RH \Rightarrow ROOH + \dot{R}$$

$$R\dot{O}_2 \Rightarrow R'CHO + R''\dot{O}$$

Degenerate Branching

$$ROOH \Rightarrow R\dot{O} + \dot{O}H$$

$$R'CHO + O_2 \Rightarrow R'\dot{C}O + H\dot{O}_2$$

Cetane Rating

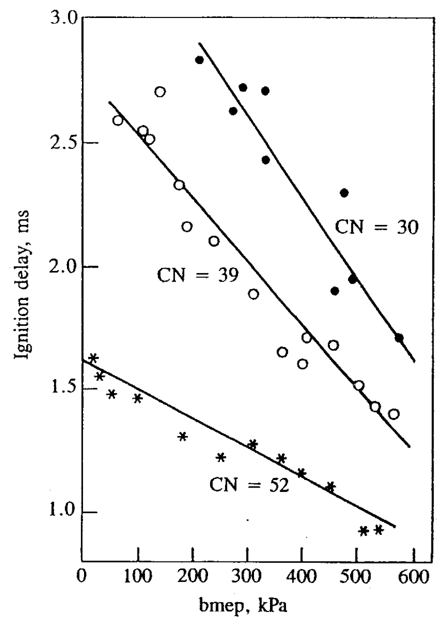
(Procedure is similar to Octane Rating for SI Engine; for details, see10.6.2 of text)

Primary Reference Fuels:

- Normal cetane ($C_{16}H_{34}$): CN = 100
- Hepta-Methyl-Nonane (HMN; $C_{16}H_{34}$): CN = 15
  (2-2-4-4-6-8-8 Heptamethylnonane)

Rating:

- Operate CFR engine at 900 rpm with fuel
- Injection at 13° BTC
- Adjust compression ratio until ignition at TDC
- Replace fuel by reference fuel blend and change blend proportion to get same ignition point
- CN = % n-cetane + 0.15 x % HMN

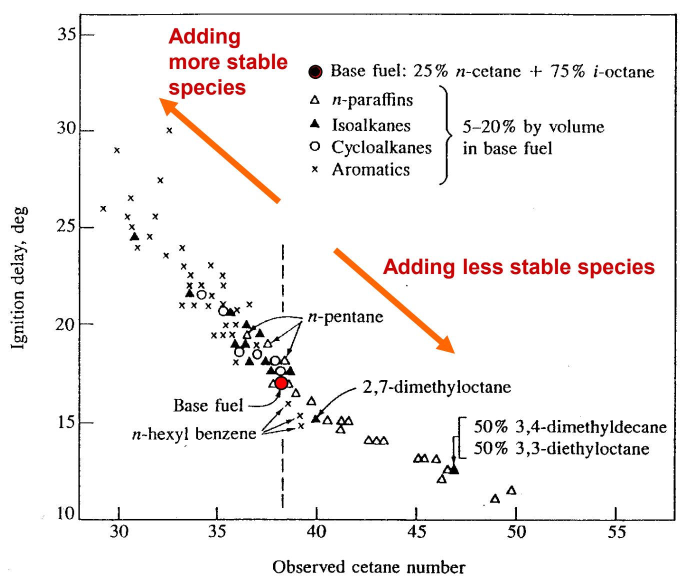


Ignition Delay

Ignition delays measured in a small four-stroke cycle DI diesel engine with r_c=16.5, as a function of load at 1980 rpm, at various cetane number

(Fig. 10-36)

Fuel effects on Cetane Number (Fig. 10-40)



Ignition Delay Calculations

- Difficulty: do not know local conditions (species concentration and temperature) to apply kinetics information

Two practical approaches:

- Use an "instantaneous" delay expression
  $\tau(T,P) = P^{-n}\exp(-E_A/T)$
  and solve ignition delay (τ_{id}) from
  $$1 = \int_{t_{si}}^{t_{si}+\tau_{id}} \frac{1}{\tau(T(t),P(t))}\,dt$$
- Use empirical correlation of τ_{id} based on T, P at an appropriate charge condition; e.g. Eq. (10.37 of text)

$$\tau_{id}(CA) = (0.36 + 0.22\bar{S}_p(m/s))\exp\left[E_A\left(\frac{1}{\tilde{R}T(K)} - \frac{1}{17190}\right) + \left(\frac{21.2}{P(bar) - 12.4}\right)^{0.63}\right]$$

E_A (Joules per mole) = 618,840 / (CN+25)

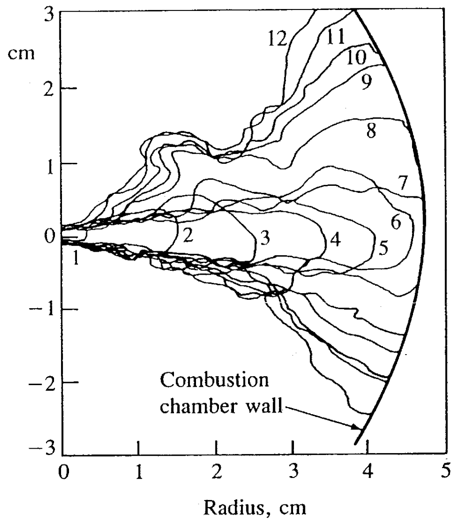
Diesel Engine Combustion
Air Fuel Mixing Process

- Importance of air utilization
  - Smoke-limit A/F ~ 20
- Fuel jet momentum / wall interaction has a larger influence on the early part of the combustion process
- Charge motion impacts the later part of the combustion process (after end-of-injection)

CHARGE MOTION CONTROL

- Intake created motion: swirl, etc.
  - Not effective for low speed large engine
- Piston created motion - squish

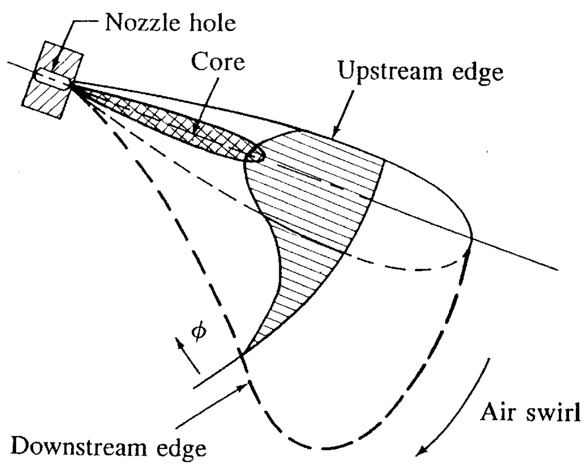
Interaction of fuel jet and the chamber wall



Sketches of outer vapor boundary of diesel fuel spray from 12 successive frames (0.14 ms apart) of high-speed shadowgraph movie. Injection pressure at 60 MPa.

Fig. 10-21

Interaction of fuel jet with air swirl

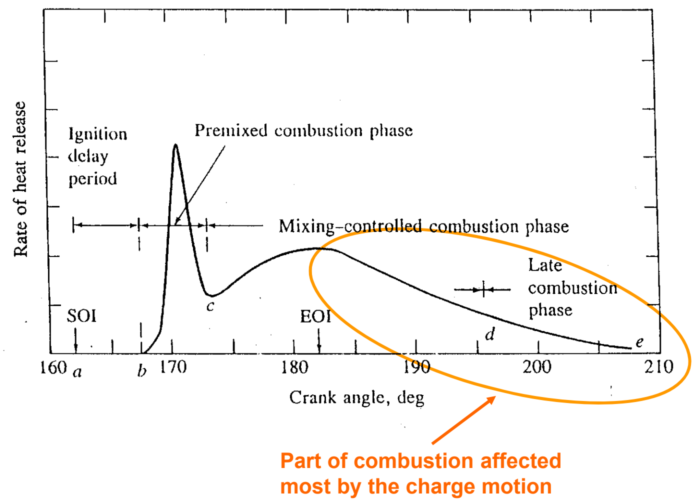


Schematic of fuel jet – air swirl interaction; Φ is the fuel equivalence ratio distribution

Fig. 10-22

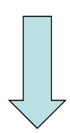
Rate of Heat Release in Diesel Combustion

(Fig. 10.8 of Text)



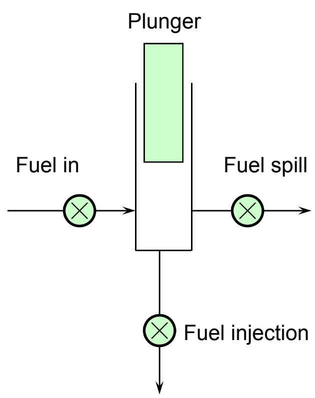
DIESEL FUEL INJECTION HARDWARE

- High pressure system
  - precision parts for flow control
- Fast action
  - high power movements



Expensive system

FUEL METERING AND INJECTION SYSTEM - CONCEPT

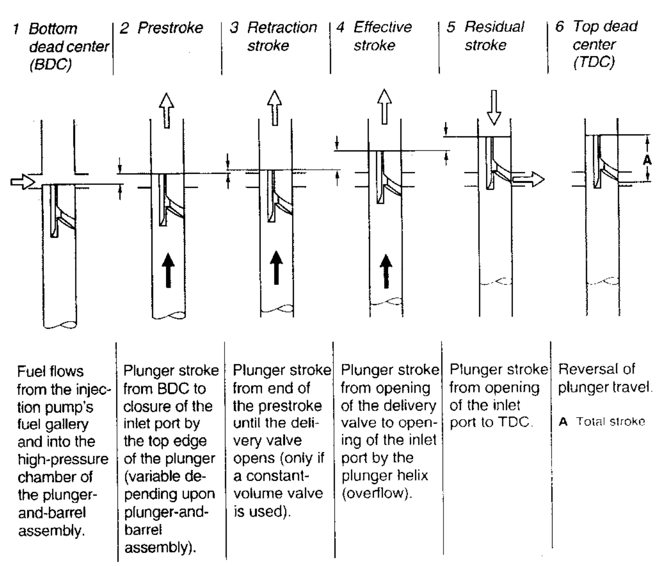


Process:

- Fill
- Pressurize
- Inject
- Spill

Fuel Delivery Control

Fig. 3: Plunger-stroke phases



From *Diesel Fuel Injection*, Robert Bosch GmbH, 1994

Fuel Rack and In-line Pump

From *Diesel Fuel Injection*, Robert Bosch GmbH, 1994

Fig. 4: Fuel-delivery control
Using a toothed control rack. a) Zero delivery. b) Partial delivery. c) Maximum delivery.
1 Pump barrel, 2 Inlet port, 3 Pump plunger, 4 Helix, 5 Control rack.

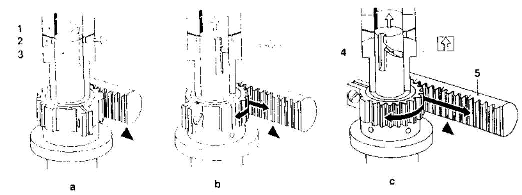
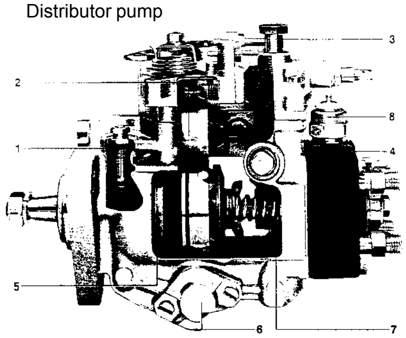
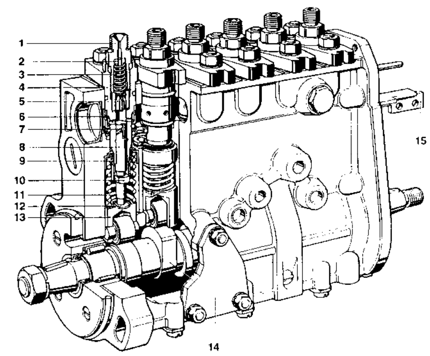


Fig. 1: PES in-line fuel-injection pump
1 Delivery-valve holder, 2 Filler piece, 3 Delivery-valve spring, 4 Pump barrel, 5 Delivery valve, 6 Inlet port and spill port, 7 Control helix, 8 Pump plunger, 9 Control sleeve, 10 Plunger control arm, 11 Plunger spring, 12 Spring seat, 13 Roller tappet, 14 Cam, 15 Control rack.

Distributor pump





Diesel Injector

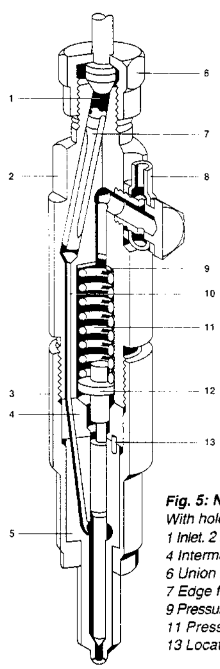
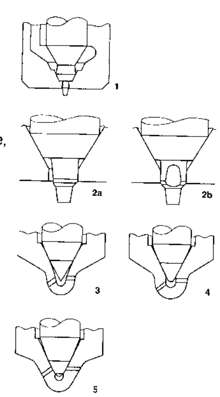
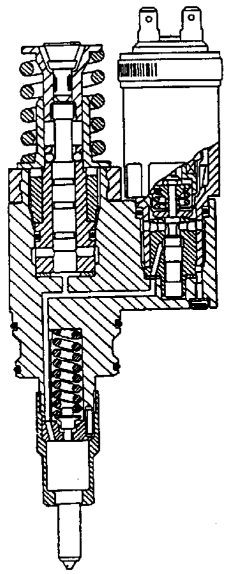


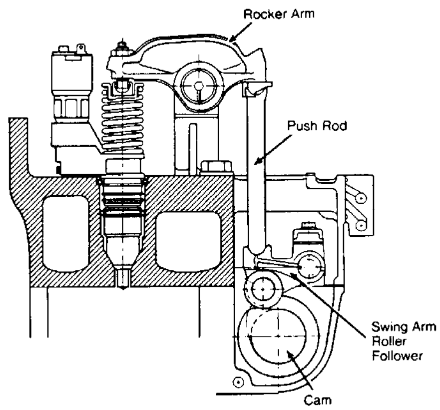
Fig. 5: Nozzle-and-holder assembly
With hole-type nozzle.
1 Inlet, 2 Nozzle-holder body, 3 Nozzle-retaining nut,
4 Intermediate element, 5 Injection nozzle,
6 Union nut with high-pressure line,
7 Edge filter, 8 Leak-off connection,
9 Pressure-adjusting shims, 10 Pressure passage,
11 Pressure spring, 12 Pressure pin,
13 Locating pins.

Fig. 2: Nozzle shapes
1 Throttling pintle nozzle,
2 Throttling pintle nozzle with flat-cut pintle,
2a Side view, 2b Front view,
3 Hole-type nozzle with conical blind hole,
4 Hole-type nozzle with cylindrical blind hole,
5 Seat-hole nozzle.



Electronic Unit Injector



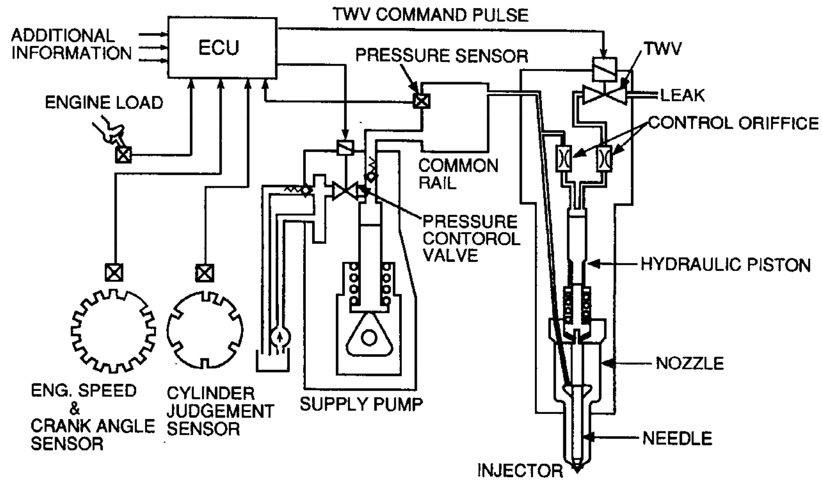


SAE Paper 891001

Injection pressure

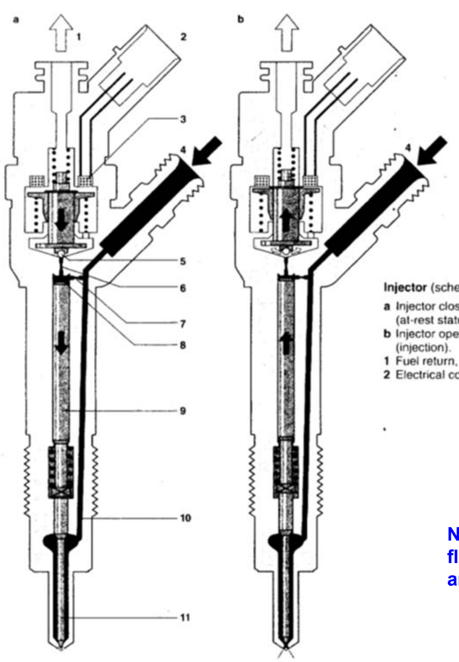
- Positive displacement injection system
  - Injection pressure adjusted to accommodate plunger motion
  - Injection pressure $\propto$ rpm^2
- Injection characteristics speed dependent
  - Injection pressure too high at high rpm
  - Injection pressure too low at low rpm

Common Rail Fuel Injection System



SAE Paper 1999-01-0833

Common Rail Injector



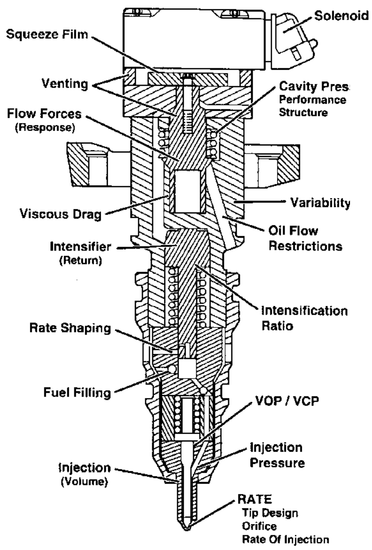
Injector (schematic)

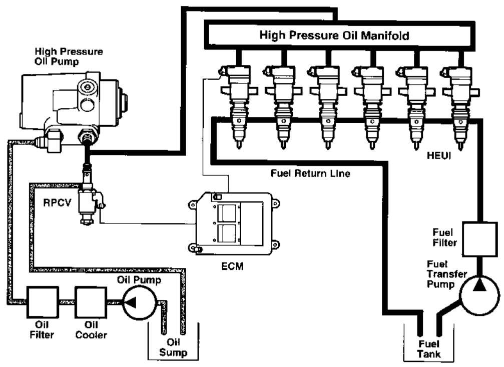
a Injector closed (at-rest status),
b Injector opened (injection).
1 Fuel return,
2 Electrical connection,
3 Triggering element (solenoid valve),
4 Fuel inlet (high pressure) from the rail,
5 Valve ball,
6 Bleed orifice,
7 Feed orifice,
8 Valve control chamber,
9 Valve control plunger,
10 Feed passage to the nozzle,
11 Nozzle needle.

Nozzle opening speed controlled by the flow rate difference between the Bleed (6) and Feed (7) orifices

From Bosch: Diesel Engine Management

Caterpillar Hydraulic Electronic Unit Injector (HEUI)

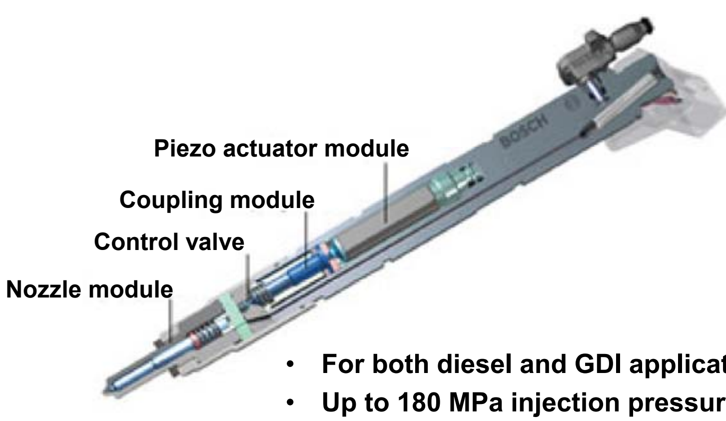




Fuel line: 200kPa; Low pressure oil: 300 kPa;
High pressure oil: up to 23 MPa;
Intensifier area ratio 7:1
Injection pressure up to 150 MPa

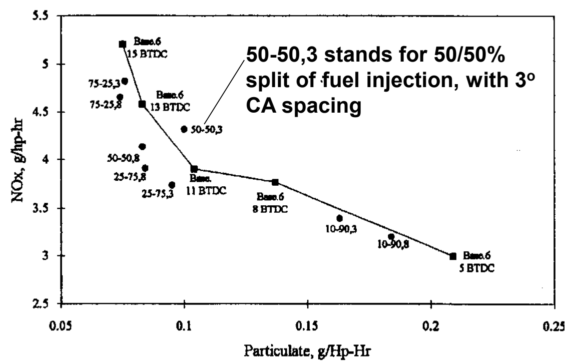
SAE Papers 930270, 930271

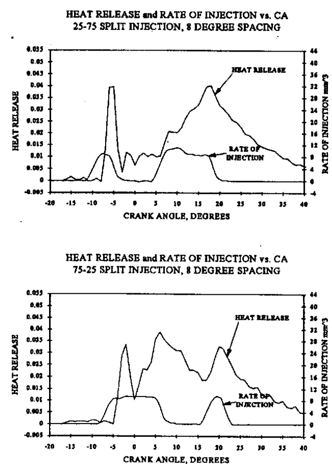
Piezoelectric injectors



- **For both diesel and GDI applications**
- **Up to 180 MPa injection pressure**
- **5 injections per cycle**
- **In vehicle production already**
- **Suppliers: Bosch; Delphi; Denso; Siemens; …**

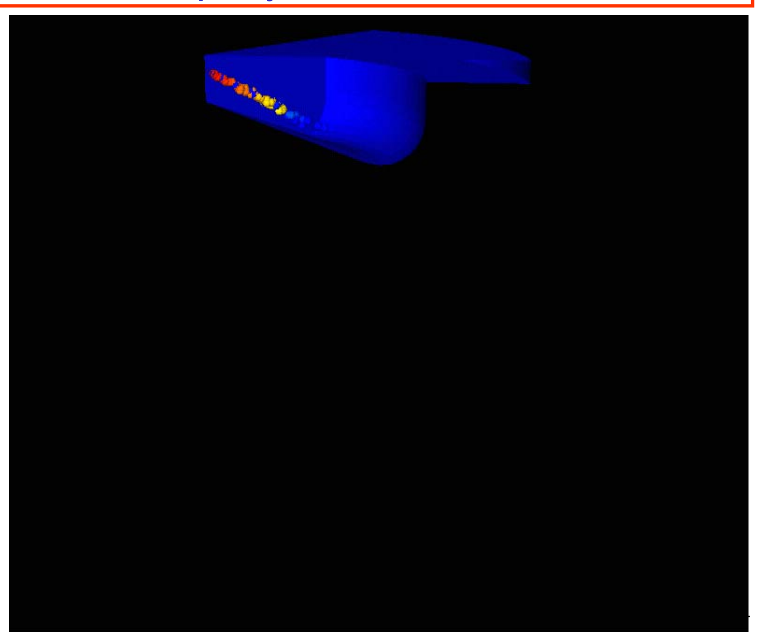
Split Injection (SAE Paper 940668)





1600 rpm, 184 KPa manifold pressure, overall fuel equivalence ratio = 0.45;

Split injection cfd simulation



CHALLENGES IN DIESEL COMBUSTION

Heavy Duty Diesel Engines

- NOx emission
- Particulate emission
- Power density
- Noise

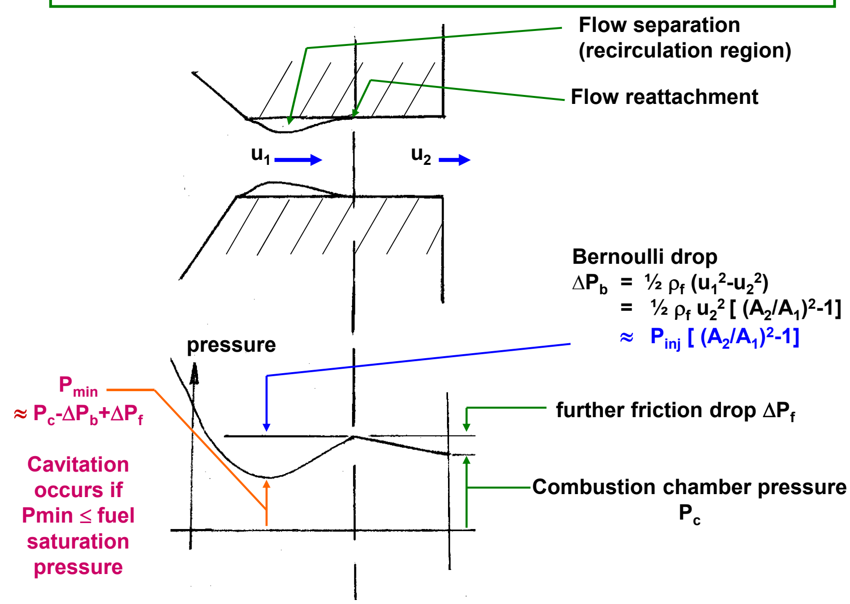
High Speed Passenger Car Diesel Engines

- All of the above, plus
  - Fast burn rate

Cavitation in Injection Nozzle

- Cavitation happens when local pressure is lower than the fluid vapor pressure
- Effects
  - Discharge rate
  - Affects the spray angle
  - Damage to the nozzle passage
- Factors affecting cavitation
  - Combustion chamber pressure
  - Local streamline curvature within the nozzle

Flow process that leads to cavitation



REFERENCES

1. **Wai Cheng.** *Internal Combustion Engines*. Massachusetts Institute of Technology: MIT Open Course Ware.
2. **Heywood, J.** *Internal Combustion Engine Fundamentals*, McGraw-Hill, New York, 1988.
3. **Pulkrabek, W.C.** *Engineering Fundamentals of the Internal Combustion Engine*, Prentice Hall, Upper Saddle River, New Jersey, 2003.
4. **Colin R. Ferguson and Allan T. Kirkpatrick**. *Internal Combustion Engines: Applied Thermal Sciences*, 2nd Edition,, John Wiley and Sons, New York, 2000.
5. **Gupta, H. N**. *Fundamentals of Internal Combustion Engines*, PHI Learning Private Limited, New Delhi, 2009.
6. **Awaludin, W. Panuntun, W.S. Alam, N. Sinaga**. *Selection of Diesel Generator for Biogas Power Plant Systems*, National Seminar on Chemical Engineering, Department of Chemical Engineering FT Undip, 2003.
7. **Sinaga, Nazaruddin**. *Design of Biogas-Air Mixer for Dual Fuel Diesel-Biogas Engines*, Journal Teknik, Year XXV, Issue I, 2005.
8. **Sinaga, Nazaruddin**. *Analysis and Engine Selection for Dual Fuel Diesel-Biogas*, Journal Rotasi, Mechanical Engineering Department, Diponegoro University, Vol. 7 No. 2, April 2005.
9. **Sinaga, Nazaruddin**. *Design of Conversion Kit for Dual Fuel Diesel-Biogas Engine Modification*, National Journal of Efficiency and Energy Conservation, Mechanical Engineering Department, Diponegoro University, Vol. 1 No. 1, September 2005.
10. **Sinaga, Nazaruddin**. *Opportunity and Strategy for Energy Saving in the Transportation Sector in Indonesia*, Proceedings, National Seminar on Energy Efficiency and Conservation (FISERGI) 2005, Diponegoro University, ISSN 1907-0063, December 2005.
11. **Cahyono, Sukmaji Indro, Gwang-Hwan Choe, and N. Sinaga**. *Numerical Analysis of a Water Brake Dynamometer Using Computational Fluid Dynamic Software*, Proceedings of the Korean Solar Energy Society Conference, 2009.
12. **Sinaga, Nazaruddin**. *The Influence of Turbulence and Pressure-Velocity Coupling Algorithm on the Simulation Results of the Flow Through the Suction Valve of the Motor Cycle Engine*, Journal of Rotation, Volume 12, No. 2, ISSN: 1411-027X, April 2010.
13. **Priangkoso, Tabah and N. Sinaga**. *Review of Fuel Consumption Mechanistic Models to be Applied on the Smart Driving Simulator Program*, Proceedings, 2nd National Science and Technology Seminar, Faculty of Engineering, Wahid Hasyim University, Semarang, June 2011.
14. **Mrihardjono, Juli and N. Sinaga**. *Driving Cycle Tests of Honda City Passenger Cars Fueled by Premium Gasoline*, Journal of Gema Teknologi, Volume 16, No. 3, October 2011, ISSN: 0852 0232.
15. **Sinaga, Nazaruddin and Tabah Priangkoso**. *Review of Empirical Models of Vehicle Fuel Consumption*, Journal of Momentum, Vol. 7, No. 1, April 2011.

16. **Supriyo and N. Sinaga**. *Design of Cooling Power of Eddy Current Dynamometer*, Journal of Eksergi, Politeknik Negeri Semarang, Vol. 7, No. 3, ISSN: 0216-8685, September 2011.
17. **Supriyo and N. Sinaga**. *Design of 250 kW Eddy Current Dynamometer*, Journal of Eksergi, Vol. 7, No. 3, ISSN: 0216-8685, September 2011.
18. **Sinaga, Nazaruddin**. *Energy-Saving Tests of Passenger Cars to Support the Smart Driving Program in Indonesia*, Proceedings, 10th National Seminar on Mechanical Engineering (SNTTM X), Mechanical Engineering Department, Faculty of Engineering, Brawijaya University, Malang, November 2011.
19. **Sinaga, Nazaruddin, T. Priangkoso, D. Widayana, and K. Abdurrohman**. *Experimental Study on the Effect of Driving Parameters on Fuel Consumption of 1500-2000 CC Passenger Cars*, Proceedings, 10th National Seminar on Mechanical Engineering (SNTTM X), Mechanical Engineering Department, Faculty of Engineering, Brawijaya University, Malang, November 2011.
20. **Sinaga, Nazaruddin and B. Prasetiyo**. *Experimental Study on the Characteristics of an Eddy Current Chassis Dynamometer*, Journal of Eksergi, Politeknik Negeri Semarang, Vol. 8, No. 2, May 2012, ISSN: 0216-8685.
21. **Sinaga, Nazaruddin and A. Dewangga**. *Tests and Preparation of Water Brake Chassis Dynamometer User Manuals*, Journal of Rotation, Vol. 14, No. 3, July 2012, ISSN: 1411-027X.
22. **Sinaga, Nazaruddin**. *Smart Driving: Fuel Saving, Emission Quality Enhancement and Accident Reduction*, Paper presented in the Seminar of Astra-Undip, Mechanical Engineering Department, Diponegoro University, November 2012.
23. **Sinaga, Nazaruddin, and Mulyono**. *Experimental Study on the Impact of Pertamax and Pertamax-Plus Fuels on the Exhaust Emissions of Motorcycles*, Proceedings, National Seminar of Research and Community Service Institution, Politeknik Negeri Semarang, 2013, ISBN: 978-979-3514-66-6, Pages 168-172.
24. **Sinaga, Nazaruddin and S. J. Purnomo**. *Relationship of Throttle Position, Engine Rotation and Gear Position on Fuel Consumption of Passenger Cars*, Eksergi, Energy Engineering Journal, State Polytechnic Semarang, Vol. 9 No. 1, January 2013.
25. **Sinaga, Nazaruddin**. *Smart Driving Training to Reduce Greenhouse Gas Emissions and Transportation Costs of Land Transportation*, Proceeding, 12th National Seminar on Mechanical Engineering (SNTTM XII), Faculty of Engineering, University of Lampung, October 2013.
26. **Sinaga, Nazaruddin, S. J. Purnomo, and A. Dewangga**. *Development of Efficient Fuel Consumption Equation Models for EFI Gasoline Fuel Passenger Cars*, Proceeding, 10th National Seminar on Mechanical Engineering (SNTTM XII), Faculty of Engineering, University of Lampung, October 2013.
27. **Sinaga, Nazaruddin, and Y. N. Rohmat**. *Comparison of the Performance of LPG and Gasoline Motorcycles*, Proceedings, National Seminar on Green Industry Technology, Center for Industrial Pollution Prevention Technology (BBTPPI) Semarang, Ministry of Industry, Semarang May 21, 2014.
28. **Syachrullah, L.I, dan N. Sinaga**. *Optimization and Prediction of Motorcycle Injection System Performance with Feed-Forward Back-Propagation Method Artificial Neural Network*, Proceedings, 2nd National Seminar on Development of Research and

Technology in Industry, Faculty of Engineering, Gajah Mada University Yogyakarta, June 2014.

29. **Paridawati and N. Sinaga**. *Reducing Fuel Consumption of an Injection System Motorcycle Using Artificial Neural Network Optimization Method with Back-Propagation Algorithm*, Proceedings, 2nd National Seminar on Development of Research and Technology in Industry, Faculty of Engineering, Gajah Mada University Yogyakarta, June 2014.
30. **M. Rifal and N. Sinaga**. *Impact of Methanol-Gasoline Blend on Fuel Consumption and Exhaust Emission of an SI Engine*, Proceedings, The 3rd International Conference on Advanced Materials Science and Technology (ICAMST 2015), Semarang State University, April 2015.
31. **Sinaga, Nazaruddin, and Mulyono**. *Experimental Study on the Motorcycle Performance with Variation of Gasoline Types*, Journal of Eksergi, Vol. 11, No. 1, ISSN: 0216-8685, Pages 1- 6, January 2015.
32. **Syachrullah, L.I, and N. Sinaga**. *Optimization and Prediction of Motorcycle Injection System Performance with Feed-Forward Back-Propagation Method Artificial Neural Network*, American Journal of Engineering and Applied Science, Vol. 8 Issue 2, pp. 236-250, ISSN: 1941-7039, February 26, 2016.
33. **Rojak, Amirur and N. Sinaga**. *Analysis of Air and Fuel Consumption on Passenger Cars Fuel with LGV*, Journal of Politeknosains, Vol. XV, No. 1, ISSN: 1829-6181, March 2016.
34. **Khudhoibi and N. Sinaga**. *Effect of Engine Remap on LGV-Fueled Car Operations*, Journal of Momentum, Islamic University of Wachid Hasyim, Vol. 12, No. 1, ISSN: 0216-7395, April 2016.
35. **Rifal, Mohamad and N. Sinaga**. *Impact of Methanol-Gasoline Fuel Blend on Fuel Consumption and Exhaust Emission of SI Engine*, AIP Conf. Proc. 1725, 020070-1–020070-6; Published by AIP Publishing, 978-0-7354-1372-6, March 2016.
36. **Sinaga, Nazaruddin and D. Alcita**. *Comparison of Fuel Consumption on EFI Car Fueled with Gasoline and Methanol-Gasoline M15*, Eksergi, Energy Engineering Journal, State Polytechnic Semarang, Polines, Vol. 12 No. 3, September 2016.
37. **Nazaruddin Sinaga**. *Preliminary Design of a Simple LPG Converter Kit for Small Scale Gasoline Engines*, Journal of Eksergi, Journal of Energy Engineering Polines, Vol. 13, No. 1, January 2017.
38. **Nazaruddin Sinaga**. *Numerical Jet-Swirling Analysis on Annulus Channels Flow Using Finite Volume Method*, Journal of Rotation, Mech. Eng. Dept., Diponegoro University, Vol. 19, No. 2, April 2017.
39. **Nazaruddin Sinaga and M. Rifal**. *Effect of Methanol-Gasoline Fuel Composition on Torque and Power of a 1200 CC EFI Passenger Car*, Journal of Rotation Vol. 19, No. 3, July 2017.
40. **Nazaruddin Sinaga**. *Design and Manufacturing of Simple Data Loggers for Motorcycle Chassis Dynamometers*, Journal of Rotasi, Vol. 20, No. 1, January 2018.
41. **Rifal, Mohamad and N. Sinaga**. Experimental Study of Methanol – Gasoline Ratio on Fuel Consumption, Exhaust Emission, Engine Torque and Power, Gorontalo Journal of Infrastructure and Science Engineering, Vol 1 (1), April 2018, pp. 47-54.

42. **Nugroho, A., Sinaga, N., Haryanto, I**. *Performance of a Compression Ignition Engine Four Strokes Four Cylinders on Dual Fuel (Diesel-LPG)*, Proceeding, The 17th International Conference on Ion Sources, Vol. 2014, 2018, 21 September 2018, AIP Publishing.

43. **Nazaruddin Sinaga, B. Yunianto, Syaiful, and W.H. Mitra Kusuma**. *Effect of Addition of 1,2 Propylene Glycol Composition on Power and Torque of an EFI Passenger Car Fueled with Methanol-Gasoline M15*, Proceeding of International Conference on Advance of Mechanical Engineering Research and Application (ICOMERA 2018), Malang, October 2018.

44. **Nazaruddin Sinaga, Syaiful, B. Yunianto, M. Rifal**. *Experimental and Computational Study on Heat Transfer of a 150 KW Air Cooled Eddy Current Dynamometer*, Proc. The 2019 Conference on Fundamental and Applied Science for Advanced Technology (Confast 2019), Yogyakarta, January 21, 2019.

45. **Nazaruddin Sinaga**. *CFD Simulation of the Width and Angle of the Rotor Blade on the Air Flow Rate of a 350 kW Air-Cooled Eddy Current Dynamometer*, Proc. The 2019 Conference on Fundamental and Applied Science for Advanced Technology (Confast 2019), Yogyakarta, January 21, 2019.

46. **Ahmad Faoji, Syaiful Laila, Nazaruddin Sinaga**. *Consumption and Smoke Emission of Direct Injection Diesel Engine Fueled by Diesel and Jatropha Oil Blend with Cold EGR System*, Proc. The 2019 Conference on Fundamental and Applied Science for Advanced Technology (Confast 2019), Yogyakarta, January 21, 2019.

47. **Johan Firmansyah, Syaiful Laila, Nazaruddin Sinaga**. *Effect of Water Content in Methanol on the Performance and Smoke Emissions of Direct Injection Diesel Engines Fueled by Diesel Fuel and Jatropha Oil Blends with EGR System*, Proc. The 2019 Conference on Fundamental and Applied Science for Advanced Technology (Confast 2019), Yogyakarta, January 21, 2019.

48. **Sinaga, Nazaruddin, M. Mel, D.A Purba, Syaiful, and Paridawati**. *Comparative Study of the Performance and Economic Value of a Small Engine Fueled with B20 and B20-LPG as an Effort to Reduce the Operating Cost of Diesel Engines in Remote Areas*, Joint Conference of 6th Annual Conference on Industrial and System Engineering (6th International Conference of Risk Management as an Interdisciplinary Approach (1st ICRMIA) 2019 on April 23-24, 2019 in Semarang, Central Java, Indonesia.

49. **Sinaga, Nazaruddin, B. Yunianto, D.A Purba, Syaiful and A. Nugroho**. *Design and Manufacture of a Low-Cost Data Acquisition Based Measurement System for Dual Fuel Engine Researches*, Joint Conference of 6th Annual Conference on Industrial and System Engineering (6th International Conference of Risk Management as an Interdisciplinary Approach (1st ICRMIA) 2019 on April 23-24, 2019 in Semarang, Central Java, Indonesia.

50. **Y Prayogi, Syaiful**, **and N Sinaga**. *Performance and Exhaust Gas Emission of Gasoline Engine Fueled by Gasoline, Acetone and Wet Methanol Blends*, International Conference on Technology and Vocational Teacher (ICTVT-2018), IOP Conf. Series: Materials Science and Engineering 535 (2019) 012013 doi:10.1088/1757-899X/535/1/012013